Alternative Fluorocarbons Environmental Acceptability Study

Refrigeration and Air Conditioning Technology Workshop
Breckenridge Hilton, Breckenridge, CO
June 23-25, 1993

THERMOACOUSTIC REFRIGERATION

Steven L. Garrett, Thomas J. Hofler, and David K. Perkins
Physics Department and Space Systems Academic Group
Naval Postgraduate School, Monterey, CA 93943

1.0 TECHNOLOGY DESCRIPTION

1.1 Historical Perspective. The thermoacoustic heat pumping cycle is the youngest technology that will be presented at this workshop. Although the reverse process - the generation of sound by an imposed temperature gradient - had been observed for several centuries by glassblowers[1] and for decades by cryogenic researchers[2]; the recognition that useful amounts of heat could be pumped against a substantial temperature gradient with a coefficient-of-performance which is a significant fraction of the Carnot limit was only made ten years ago[3], with the first demonstration, including efficiency measurements, being made in 1986[4].

This discovery was made even more significant by the recognition that the thermoacoustic heat pumping cycle was intrinsically irreversible. Traditional heat engine cycles, such as the Carnot Cycle typically studied in elementary thermodynamics courses, assume that the individual steps in the cycle are reversible. In thermoacoustic engines, the irreversibility due to the imperfect (diffusive) thermal contact between the acoustically oscillating working fluid and a stationary second thermodynamic medium (the "stack") provides the required phasing. This "natural phasing"[4] has produced heat engines which require no moving parts other than the self-maintained oscillations of the working fluid.

During this relatively short period, several refrigerators and prime movers have been fabricated and tested at Los Alamos National Laboratories[3-5] and two refrigerators for spacecraft applications were built at the Naval Postgraduate School. The Space ThermoAcoustic Refrigerator[7] was flown on the Space Shuttle *Discovery* (STS-42) in January, 1992, and the ThermoAcoustic Life Sciences Refrigerator (TALSR)[8] is now being tested and should be characterized completely by October, 1993.

TALSR was designed to pump 700 Btu/hr in the refrigerator mode (+4°C) and 400 Btu/hr in the freezer mode (-22°C). This makes it the first thermoacoustic refrigerator which would be capable of operation as a conventional domestic food refrigerator/freezer. At the present time, there are several preliminary designs which should be capable of one-half ton to three tons of air conditioning capacity, but no prototypes are currently under construction.

1.2 A Simple Invisid Model of the Thermoacoustic Heat Pumping Process. Although a complete and detailed analysis of the thermoacoustic heat pumping process is well beyond the scope of this paper, the following simple, invisid, Lagrangian representation of the cycle contains the essence of the process. A complete analysis[6] would necessarily include the gas viscosity, finite wavelength effects, longitudinal thermal conduction along the stationary second thermodynamic medium and through the gas, and the ratio of the gas and solid dynamic heat capacities.

A schematic diagram of a simple, one-quarter wavelength thermoacoustic refrigerator is shown in Figure 1.

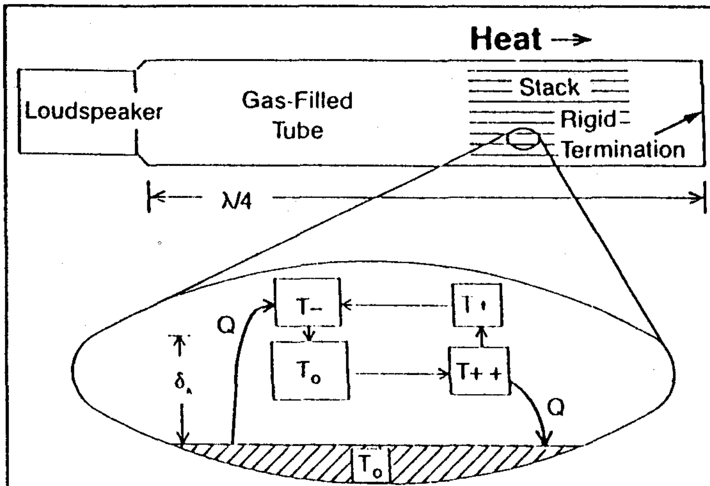

Figure 1. Schematic diagram of a one-quarter wavelength thermoacoustic refrigerator.

The thermal penetration depth, δ_κ, represents the distance over which heat will diffuse during a time which is on the order of an acoustic period, T = 1/f, where f is the acoustic frequency. It is defined[9] in terms of the thermal conductivity of the gas, κ, the gas density, ρ, and its isobaric specific heat (per unit mass), c_p.

$$\delta_\kappa = \sqrt{\frac{\kappa}{\pi f \rho c_p}} \tag{1}$$

This length scale is crucial to understanding the performance of the thermoacoustic cycle since the diffusive heat transport between the gas and the "stack" is only significant within this region. It is for that reason that the stack and the spacing between its plates are central to the thermoacoustic cycle.

For this analysis we will focus our attention on a small portion of a single plate surface within the "stack" and the adjacent gas which is undergoing acoustic oscillations. The distance from the solid stack material is small enough that a substantial amount of thermal conduction can take place in an amount of time which is on the order of the acoustic period. In the lower half of Figure 1, a small portion of the stack has been magnified and a parcel of gas undergoing an acoustic oscillation is shown. The four steps in the cycle are represented by the four boxes which are shown as moving in a rectangular path for clarity. In reality they simply oscillate back and forth. As the fluid oscillates back and forth along the plate, it undergoes changes in temperature due to the adiabatic compression and expansion resulting from the pressure variations which accompany the standing sound wave. The compressions and expansions of the gas which constitute the sound wave are adiabatic if they occur far from the surface of the plate. The relation between the change in gas pressure due to the sound wave, p_1, relative to the mean (ambient) pressure, p_m, and the adiabatic temperature change of the gas, T_1, due to the acoustic pressure change, relative to the mean absolute (Kelvin) temperature, T_m, is given below in equation (2).

$$\frac{T_1}{T_m} = \frac{\gamma - 1}{\gamma} \frac{p_1}{p_m} \tag{2}$$

Although the oscillations in an acoustic heat pump are sinusoidal functions of time, Figure 1 depicts the motion as articulated (a square wave) in order to simplify the explanation. The plate is assumed to have a mean temperature , T_m, and a temperature gradient,∇T, referenced to the mean position, $x = 0$. The temperature of the plate at the left-most position of the gas parcels excursion is therefore $T_m - x_1\nabla T$, and at the right-most excursion is $T_m + x_1\nabla T$.

In the first step of this four-step cycle, the fluid is transported along the plate by a distance $2x_1$ and is heated by adiabatic compression from a temperature of $T_m - x_1\nabla T$ to $T_m - x_1\nabla T + 2T_1$. The adiabatic gas law provides the relationship between the change in gas pressure, p_1, and the associated change in temperature, T_1, as described in equation (2). Because we are considering a heat pump, work, in the form of sound, was done on the gas parcel hence it is now a temperature which is higher than that of the plate at its present location (*i.e.*, $|x_1\nabla T| < |T_1|$).

In the second step, the warmer gas parcel transfers an amount of heat, dQ_{hot}, to the plate by thermal conduction at constant pressure and its temperature decreases to that of the plate, $T_m + x_1\nabla T$. In the third step, the fluid is transported back along the plate to position $-x_1$ and is cooled by adiabatic expansion to a temperature $T_m + x_1\nabla T - 2T_1$. This temperature is lower than the original temperature at location $-x_1$, so in the fourth step the gas parcel adsorbs an amount of heat, dQ_{cold}, from the plate thereby raising its temperature back to its original value, $T_m - x_1\nabla T$.

The net effect of this process is that the system has completed a cycle which has returned it to its original state and an amount of heat, dQ_{cold}, has been transported up a temperature gradient by work done in the form of sound. It should be stressed again that no mechanical devices were used to provide the proper phasing between the mechanical motion and the thermal effects.

If we now consider the full length of the stack as shown in the upper portion of Figure 1, the overall heat pumping process is analogous to a "bucket brigade" in which each set of gas parcels picks up heat from its neighbor to the left at a lower temperature and hands off the heat to its neighbor to the right at a higher temperature. Heat exchangers are placed at the ends of the stack to absorb the useful heat load at the left-hand (cold) end of the stack and exhaust the heat plus work (enthalpy) at the right-hand (hot) end of the stack. The fact that the gas parcels actually move a distance which has typically been on the order of several millimeters means that intimate physical contact between the heat exchangers and the stack is not crucial.

2.0 APPLICATIONS

The applications of thermoacoustic engines fall into two categories which depend upon whether the refrigerator is powered by electricity or by heat. Although the heat driven thermoacoustic refrigerators and cryocoolers are attractive for applications where there is abundant heat or waste heat, at the present time, only two thermoacoustically driven refrigerators have been demonstrated. The first was a "beer cooler"[5,10] and the second was a thermoacoustically driven orifice-pulse-tube cryocooler designated the "Coolahoop"[11]. A more compact commercial version of the Coolahoop is now under development for cooling of high speed electronics. Several other heat-driven thermoacoustic refrigerators are currently in the design stages for the above applications including a refrigerator for storage of medical supplies and vaccines in Bangladesh, a solar driven refrigerated cargo container for transportation of tropical fruits, and a natural gas liquefaction plant.

Work on electrically powered thermoacoustic refrigeration has, until last year, been concentrated on laboratory experiments and spacecraft applications. At the present time, Ford Motor Company is developing thermoacoustic refrigerators for proprietary applications. NPS is currently developing two refrigerators. One is a third-generation, single-stage thermoacoustic cryocooler (TAR-3) which is designed to reach high-T_c superconductor transition temperatures. The other is TALSR, which is capable of producing cooling comparable to commercial domestic refrigerator/freezers. TALSR was also designed for use on-board the Space Shuttle[8]. The first commercial application of a TALSR-like design, which will use a less expensive driver, will be targeted to a "niche" market which we are unwilling to disclose at this time.

Due to the simplicity of its operation and the use of only one moving part, thermoacoustic refrigeration is also be suitable for cooling the latest generation of computer chips which can run at twice their room temperature design speeds when their temperature is reduced to -50°C.

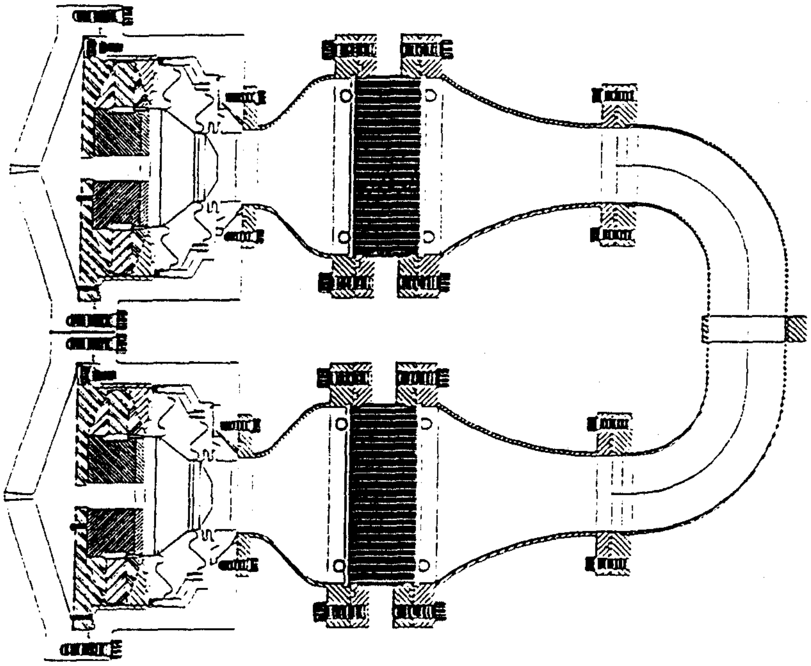

Figure 2. Cross-sectional diagram of the half-wavelength resonant TALSR. Two separate drivers are used for redundancy in space applications. A commercial unit would use a single double-acting drive. The two stacks and four fluid-filled heat exchangers are configured so that the total temperature span is greater than that of either individual stack.

3.0 BENEFITS

3.1 Inert working fluid. Helium, being an inert gas, cannot participate in chemical re actions and hence no toxicity, flammability, or negative environmental effects (ODP=GWP=0).
3.2 No sliding seals or lubrication. Due to the high frequency operation, high powers can be achieved with small displacements so no sliding seals or gas bearings are required. This also means that no "tight tolerance" machined parts are required thereby reducing manufacturing costs.
3.3 Very few simple components. Electrically driven systems require only one moving part and thermally driven systems have no moving parts. The "stack" can be fabricated from cheap plastics.
3.4 Large range of working temperatures. Depending upon the position and length of the stack in the acoustic standing wave field, one can trade off the temperature span and the heat pumping power. Different working fluids are therefore not required for different temperature ranges.
3.5 Intrinsically suited to proportional control. Just as one is able to control the volume of a stereo system, a electrically driven thermoacoustic refrigerator's cooling power is continuously variable. This allows improved overall efficiency by doing rapid cool-down at a lower COP and then maintaining heat leak losses at higher COP. This "load matching" can also reduce heat exchanger inefficiencies by minimizing temperature differences within the fluids and exchangers.
3.6 Immaturity. Thermoacoustics is the youngest of the heat engine cycles. It is more likely that important breakthroughs which substantially improve performance and manufacturability will still occur here rather than the older technologies which have already "skimmed the cream".

4.0 TECHNICAL ISSUES

4.1 Immaturity. Because thermoacoustics is the youngest of existing heat engine cycles, it lacks the infrastructure (suppliers, sales and service base, educational programs, etc.) which can enhance marketability. In addition, since there are presently no commercial products on the market, thermoacoustics does not have a "cash flow" which can be "tapped" to make either incremental component improvements or to finance general research and development efforts.

4.2 Efficiency. Although computer models[12] of TALSR predict that it will have a Coefficient-of-Performance Relative to Carnot (COPR) of 42% (exclusive of motor inefficiencies and secondary heat exchange fluid pumps), TALSR has not yet been tested. The previous thermoacoustic cryocooler designs have been optimized for temperature span rather than COP. Their best measured performance has given a COPR ≤ 20%, again exclusive of electroacoustic efficiency.

4.3 Power density. The simple boundary layer models of thermoacoustic engine performance[6,12] may not apply as acoustical amplitudes are increased. If acoustic mach numbers are restricted to $M_{ac} \leq 5\%$, then the realizable power density of conventional thermoacoustic stack geometries may be restricted to 10 Tons (35 kW) per square meter of stack cross-sectional area at working fluid pressures below 20 atm. Higher power research refrigerators and numerical hydrodynamic computer simulations would be very useful to determine what would ultimately limit the power density.

4.4 Electroacoustic conversion. Although electrical to acoustical conversion efficiencies on the order of 90% are, in principle, realizable at reasonable cost, present thermoacoustic drivers have had electroacoustic efficiencies under 50%. This should not be a problem since efficiencies for similar linear motor technology in Stirling applications as high as 93% have been measured[13].

4.5 Secondary heat transfer. All thermoacoustic engines produced thus far have used either conduction for small heat loads (<10 Watts) or electrically pumped heat exchange fluids for large heat loads (>100 Watts). Unlike the vapor compression (Rankine) cycles, the working fluid in a thermoacoustic refrigerator/chiller is not circulated outside the engine. In order to obtain maximum overall efficiency (*i.e.*, net COP), it is therefore necessary to simultaneously optimize primary and secondary heat exchanger geometry, transfer fluid thermophysical parameters, transfer fluid flow rates, and electrical pump or heat pipe performance, all subject to economic constraints, in order to achieve the best performance at the lowest cost.

4.6 The "talent bottleneck." Because thermoacoustics is a new science and requires expertise in a diverse number of non-traditional disciplines within the refrigeration and HVAC communities (acoustics, transduction, gas mixture thermophysics, PID, PLL and AGC control, etc.), there are very few experimentalists who are interested or capable of research in this field. This severely limits the number of potentially promising applications which can be pursued simultaneously.

5.0 ECONOMICS

All thermoacoustic engines which have been produced to date have been research prototypes. The costs have been typically 1-2 M$, which accounts primarily for scientific and technical staff salaries. No systematic cost projections or comparisons to existing system costs have been attempted. Limited commercialization attempts which address niche applications are expected over the next three years and should begin to provide some economic benchmarks which would lead to reliable cost estimates.

6.0 TECHNOLOGY OUTLOOK

Those of us who work with thermoacoustics feel the outlook is bright for the reasons enumerated in Section 3.0 of this paper. We recognize that our strongly positive outlook is both prejudicial and self-serving. On the other hand, the failure of technology outlook projections made by those who are not knowledgeable in thermoacoustics can be equally prejudicial, self-serving, and more importantly, wrong. This may be best illustrated by the recent analysis of "Energy Efficient Alternatives to Chlorofluorocarbons" prepared for the Department of Energy by A. D. Little, Inc. In that study[14], several domestic refrigeration technologies were ranked from 1 (Lowest) to 5 (Highest) based on the probability of success and assigned a 1-5 priority for R&D support.

In that A. D. Little analysis, Stirling Cycle was evaluated in the areas of analytical tools, linear drive systems, compact heat exchangers, reliability, and market potential of prototype designs. The sum of the scores for "probability of success" and "R&D priority" averaged 8.4 ± 0.6 out of a possible maximum sum of 10. The sum for thermoacoustics (improperly labeled Thermal Acoustic) was 2, with the minimum sum being 2! Of the other fourteen technologies evaluated in that table, none of the others had a sum lower than five.

The ultimate failure of that A. D. Little analysis can best be established by the "head-to-head" comparison that was sponsored by the Life Sciences Division of NASA. In an attempt to replace the existing Space Shuttle Life Sciences refrigerator/freezer, NASA awarded three contracts to companies with potential replacement technologies. One went to A. D. Little for a scroll pump/vapor compression technology. The other two contracts went to Sunpower, Inc., for a linear motor Stirling technology and to NPS for thermoacoustics. As of the date of this conference, less than a year after the award of the NPS contract, the A. D. Little team has dropped out and the Stirling system has been delivered with only one-third of the originally specified heat pumping capability. At this point, it appears that only the thermoacoustic technology will meet the original contract specifications.

When attempting to predict the future utility of a new discovery or emerging technology, it is always useful to recall the observations made by Prof. Faraday, D.C.L., F.R.S., in 1817[15]:

> "Before leaving this subject, I will point out the history of this substance, as an answer to those who are in the habit of saying to every new fact, 'What is the use?' Dr. Franklin says to such, 'What is the use of an infant?' The answer of the experimentalist is, 'Endeavour to make it useful.'"

We know that reports that this infant was stillborn are wrong! We feel that thermoacoustics is still too immature to make definitive technological projections. The growth curves for both efficiency and heat pumping capacity are still very steep and the number of industrial and academic researchers, though still small, is growing at an increasing rate. (Thermoacoustic cooling demonstration units are now even starting to appear in high school science fairs!) At this point, the only outlook we can guarantee is that thermoacoustic systems will continue to prove that the initially pessimistic outlook of those unfamiliar with this technology were wrong.

7.0 REFERENCES

1. J. W. Strutt (Lord Rayleigh), *The Theory of Sound*, 2nd ed., Vol. II (Dover, 1945), §322j.
2. T. Yazaki, A. Tominaga, and Y. Narahara, "Experiments on Thermally Driven Acoustic Oscillations of Gaseous Helium," J. Low Temp. Phys, **41**, 45 (1980).
3. J. C. Wheatley, T. Hofler, G. W. Swift, and A. Migliori, "Experiments with an Intrinsically Irreversible Acoustic Heat Engine," Phys. Rev. Lett. **50**, 499 (1983); "An Intrinsically Irreversible Thermoacoustic Heat Engine," J. Acoust. Soc. Am. **74**, 153 (1983); "Acoustical Heat Pumping Engine," U.S. Patent No. 4,398,398 (Aug. 16, 1983); "Intrinsically Irreversible Heat Engine," U.S. Patent No. 4,489,553 (Dec. 25, 1984).
4. T. J. Hofler, "Thermoacoustic Refrigerator Design and Performance," Ph.D. dissertation, Physics Dept., Univ. Calif. San Diego (1986); "Concepts for Thermoacoustic Refrigeration and a Practical Device," Proc. 5th Int. Cryocooler Conf. 18-19 Aug 1988, Monterey, CA; "Acoustic Cooling Engine," U. S. Patent No. 4,722,201 (Feb. 2, 1988).
5. J. C. Wheatley and A. Cox, "Natural Engines," Phys. Today **38**, 50 (1985).
6. G. W. Swift, "Thermoacoustic Engines," J. Acoust. Soc. Am. **84**(4), 1145-1180 (1988).
7. S. L. Garrett, J. A. Adeff, and T. J. Hofler, ThermoAcoustic Refrigeration for Space Applications," J. Thermophysics and Heat Transfer (AIAA) 7(3), (1993).
8. S. Garrett, "ThermoAcoustic Life Sciences Refrigerator," NASA Tech. Report No. LS-10114, Johnson Space Center, Space and Life Sciences Directorate, Houston, TX (October 30, 1991).
9. L. D. Landau and E. M. Lifshitz, *Fluid Mechanics* (Pergamon Press, 1959), §52.
10. J. C. Wheatley, G. W. Swift, A. Migliori, and T. Hofler, "Heat-driven Acoustic Cooling Engine having no Moving Parts," U. S. Patent No. 4,858,441 (Aug. 22, 1989).
11. R. R. Jones, "High-Tech Elite," R&D Magazine 32(10), 61 (1990).
12. W. Ward and G. W. Swift, "Design Environment for Linear ThermoAcoustic Engines", Los Alamos National Labs (October, 1992), preliminary release.
13. D. Berchowitz, Sunpower, Inc., at this conference.
14. A. D. Little, Inc., "Energy Efficient Alternatives to Chlorofluorocarbons," Revised Final Report-Ref. 66384 (US Dept. of Energy, ER-33, GTW), April, 1992, Table 2-2, pg. 2-8.
15. W. L. Bragg and R. Porter, *The Royal Institution Library of Science*, Physical Sciences-Vol 2, (Applied Science, Essex, England, 1970), pg. 65.